FIELD PERFORMANCE OF A GROUND-COUPLED HEAT PUMP IN ABILENE, TEXAS

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ABSTRACT

A nominal 10.6 kW (3 ton), vertical-configuration ground-coupled heat pump was installed in Abilene, Texas in December 1989 and was monitored until May 1993 using a remote data acquisition system. The unit was installed in the guest officer quarters at Dyess Air Force Base. Monitored data included: temperature and relative humidity of return and supply air, water temperature entering and leaving the condenser, power consumption of the individual system components, cycling rate, on-time, and soil temperatures at various depths and radial locations. Water and air flow rates were measured twice during the monitored period, and have remained constant. The measured quantities allow calculation of instantaneous capacity, power, coefficient of performance (COP), and ground-coil heat rejection.

Data for operation in the cooling and heating mode are discussed here. Based on the experimental data, it was discovered that the water temperature entering the condenser (EWT) exhibited a prolonged minimum after start-up due to cooling of the water during the off-cycle when operating in the cooling mode. The decreased levels of EWT early in the cycle increased capacity and decreased power, both acting to increase the COP. Seasonal COPs for heating and cooling were estimated from the data and are presented.

INTRODUCTION

Ground-coupled heat pumps (GCHPs) offer the potential for increased performance relative to air-source heat pumps (ASHPs) because of the more favorable temperature of the soil as opposed to the ambient air. In Abilene, Texas, for instance, the soil temperature at depths beyond 12 m is about 22°C year round. This temperature clearly represents an improved heat sink over 38°C air in summer, and an improved heat source over −1°C air in winter. While the potential for improved thermal performance exists, assessing whether or not it is achieved requires careful experimental monitoring. The results of one such effort are described herein.

A nominal 10.6 kW (3 ton), vertical-configuration GCHP was installed in Abilene, Texas in December 1989 and was monitored until May 1993 using a remote data acquisition system (DAS). The unit was instrumented sufficiently to allow determination of the instantaneous capacity, power, coefficient of performance (COP), ground-coil heat rejection, and cycling characteristics. Compilation of these data over the heating and cooling seasons allowed seasonal heating and cooling efficiencies to be calculated.

This paper will first discuss the experimental GCHP, the equipment used to monitor it, and the experimental uncertainties. This will be followed by a discussion of the start-up characteristics of GCHPs. Third, the success of the GCHP in achieving improved thermal performance is assessed by comparing the water temperature entering the condenser (EWT) to the outdoor air temperature. Finally, the seasonal performance of the GCHP over three cooling seasons and two heating seasons is presented.

EXPERIMENTAL SYSTEM

A nominal 10.6 kW, vertical U-tube ground-coupled heat pump was installed in guest officer's quarters of Dyess Air Force Base in Abilene, Texas in December 1989. Monitored variables included: water temperature entering and leaving the condenser, temperature and relative humidity of return and supply air, power consumption of each individual

Table 1 - Manufacturer's Predicted Cooling Performance Data

EWT (C)	Return Air (temperature/wb °C)	Capacity (kW)	Power (kW)	COP
21	27/19	13.2	2.90	4.1
21	24/17	12.2	2.84	3.8
32	27/19	10.4	3.24	3.2
32	24/17	9.6	3.17	3.0
43	27/19	9.0	3.59	2.5
43	24/17	8.3	3.51	2.4

component, cycling rate, on-time, and soil temperature at several depths and radial locations. Air flow and water flow rates were measured twice during the monitored period, and remained constant. A schematic of the GCHP and measurement points is included in Figure 1. Figure 1 depicts a domestic hot water (DHW) heat exchanger on the GCHP. In cooling, the DHW heat exchanger desuperheats the refrigerant leaving the compressor, resulting in less heat rejection to the ground-coil.

The manufacturer's predicted cooling data for the GCHP are included in Table 1. Regression equations were developed for capacity and power based on field data which were obtained when the unit was operating in a quasi-steady-state mode (on-time>18 minutes). These equations are included in the thesis of Dobson [1991], and show the manufacturer's predicted capacity data to be approximately 5% low, and the predicted power data to be 3 to 8% high.

The ground-coil consisted of 49 m long supply and return headers of nominal 3.18 cm polyethylene, and three 67 m deep U-tubes of nominal 1.9 cm polyethylene connected in parallel. The ground-coil was connected in a reverse-return configuration as indicated in Figure 2. The total length of the individual flow paths was 232 m.

The data were collected remotely with a data acquisition system (DAS) which sampled all channels at approximately four second intervals, and output all data to a time series record (TSR) at user specified intervals ranging from 1 to 30 minutes. The data in the TSR was downloaded to a personal computer via phone modem. All analog data, which included temperature and relative humidity measurements, were output to the TSR as instantaneous values. The power measurements were average values over the time interval. Instantaneous capacity and power were calculated only for intervals of one or two minutes.

All temperature measurements were made with calibrated 1000 Ω platinum resistance temperature detectors (RTDs) with an uncertainty of 0.3°C. Averaging grids of four RTDs were used in the supply and return air ducts due to the likelihood of uneven temperature distributions. Relative humidity measurements were made with capacitive type relative humidity sensors. These sensors were factory calibrated with an uncertainty of 2%. The integrity of the relative humidity measurements was verified at the beginning and end of each cooling season by comparing the supply and return air specific humidities when the fan was running without the compressor. After the fan was allowed to run long enough to dry the coil (approximately 30 minutes), the calculated specific humidities agreed to within 2%. Power inputs to all components were measured with current transformers with an uncertainty of 3%. The air flow rate was determined by measuring the temperature difference across the 10 kW bank of strip heaters, into which the power input was measured. The volumetric flow rate was computed from an energy balance assuming no heat loss to the surroundings:

$$Q = \frac{\dot{P}_{sh} + \dot{P}_f}{\rho c_p \Delta T} \quad (1)$$

This method provided an experimental uncertainty of 3.9%, and was verified to within 3% by measurements of air velocities in the cross section of the return air duct which were made with a hand held hot-wire anemometer. The adiabatic assumption was justified since the temperatures were measured directly before and after the strip heaters, resulting in a small heat transfer area.

The capacity was calculated based on the volumetric air flow rate given by Equation 1 and the measured temperature and relative humidity of the supply and return air. This is given by:

$$\dot{Q}_c = \rho Q(i_r - i_s) \quad (2)$$

In cooling operation, the enthalpy difference in Equation 2 was composed of both latent and sensible heat components, while only sensible heat components were considered in heating operation. The uncertainty in capacity was computed based on the uncertainties in air flow rate and supply and return air conditions. The calculated uncertainty was 8.8% for a 8.8 kW capacity, and 7.6% for a 10.5 kW capacity. The total power input included the power to the compressor, the fan, and the water pump:

$$\dot{P}_t = \dot{P}_{comp} + \dot{P}_f + \dot{P}_p \quad (3)$$

Including the uncertainty in the power measurement, the uncertainty in coefficient of performance (COP) ranged between 8 and 9%, depending on the capacity.

EXPERIMENTAL RESULTS

The performance of a GCHP must be addressed on several time scales to fully understand how it compares to an ASHP. Interesting differences exist between the start-up performance of GCHPs and ASHPs which require access to instantaneous performance data over the first few minutes of the cycle. This constitutes the shortest time scale on which data comparisons are necessary. Comparisons of load profiles for GCHPs and ASHPs require comparisons of EWT and air temperature over hourly periods throughout the day, constituting a second time scale. Finally, long term performance trends must be assessed by studying seasonal performance. Data on each of these time scales are presented subsequently.

Start-Up Characteristics

The start-up operation of a GCHP is driven by the value of the water temperature entering the condenser (EWT). EWT plays the same role in a GCHP that outdoor air temperature does in an ASHP. The fact that EWT varies during start-up, while the outdoor temperature of an ASHP does not, is the primary reason that GCHPs behave differently than ASHPs during start-up.

Figure 3 shows the variation of EWT with on-time for four different cooling cycles on September 3, 1990. Also included for reference are soil temperatures measured at depths of 1.5 and 30 m, both at a radial location 6 m from the center U-tube. The off-time preceding start-up varied from a low of 17 minutes to a high of 35 minutes. The trends in EWT for each of the cycles were similar. During the first cycle of water through the ground-coil, which lasted approximately 9.5 minutes, three distinct periods emerged. The first period occurred during the first 3 to 4 minutes on-time, and was characterized by rapidly decreasing EWT values. The second period occurred until about 6 minutes on-time, and was characterized by an almost constant value of EWT which was the lowest in the cycle. The third period occurred from about 6 to 9.5 minutes on-time, and exhibited rapidly increasing EWT values. These same trends occurred in subsequent cycles of the water, although the oscillations became less pronounced. The physical reason for the large temperature variations in the initial water cycle was that during the off-cycle, the water cooled towards the surrounding soil temperature. The rapid decrease in the first portion of the cycle occurred as the water in the condenser was displaced with water from deeper in the ground-coil, near cooler soil. The second period, characterized by almost uniform water temperature, occurred as the water which was in the deepest part of the coil during the off-cycle, where the soil temperature was lowest and almost constant, entered the condenser. The increase in EWT during the third period occurred as the cool water which had occupied the deep part of the U-tubes during the off-cycle was replaced with water from the shallower sections. Although the shape of the EWT versus on-time plot was similar for each of the cycles, the cycles preceded by longer off-times exhibited lower values of EWT.

The decreased values of EWT at the beginning of the on-cycle tend to increase the

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capacity and decrease the power, both increasing the COP. This suggests that cycling may increase the efficiency of a GCHP. This is contrary to the behavior of ASHPs, where cycling has been shown to have purely detrimental effects [Katipamula,1989]. To assess the extent to which the favorable values of EWT early in a cycle increased the performance over that expected in the steady state, the following nondimensional group was developed:

$$COP^{+} = \frac{COP}{COP(EWT_{ss}, i_r)} \quad (4)$$

The numerator of COP^+ is the instantaneous value of COP, while the denominator is the value of COP predicted by the regression equations from at the steady state value of EWT and the instantaneous value of return air enthalpy, i_r (Dobson, 1991). This definition quantifies the improvement in COP due to values of EWT that are below those that would be achieved at steady state.

Figure 4 is a plot of COP^+ for the same four cycles for which EWT data are shown in Figure 3. One can see from this figure that an improved COP is indeed achieved early in the cycle due to the lower values of EWT, and that this improvement is more pronounced for longer off times. The maximum value of COP^+ was about 1.13, indicating that at that point the unit was operating 13% more efficiently than it would have at steady state. The factors that kept COP^+ from rising above unity for the first minute of the cycle are believed to be the same ones that degrade ASHP start-up performance, namely thermal capacitance of the heat exchangers and refrigerant dynamics. These effects are discussed more thoroughly in Dobson et al. [1993], or in the ASHP literature [Katipamula,1989].

The trends in EWT that lead to improved start-up performance during cooling operation have a similar effect during heating operation. In heating, increasing EWT increases capacity and decreases power. During the off-cycle, the water in the ground-coil gains heat from the soil and thus it displays its maximum temperatures during the first cycle of the water through the coil. EWT data for a single heating cycle in Figure 5 illustrate this trend. This EWT profile also leads to increased COP values early in the cycle, and values of COP^+ greater than unity.

Daily Variation of EWT

Since the load on the home varied considerably over the day, one would naturally expect the value of EWT to also vary over the day. Comparing this variation in EWT to that in the outdoor air temperature provides an indication of the thermal advantage offered by the GCHP. Before this comparisons is made, however, a short discussion of the rationale behind the comparison is in order.

The argument normally proposed to justify GCHP use is that the soil temperature is favorable to the air temperature for either rejecting or extracting heat. This argument has its origins in the Carnot cycle, which assumes heat exchange processes with infinitesimal temperature differences. In reality, however, no heat exchanger has the infinite conductance required to operate in this manner. This means that comparing the air temperature to the soil temperature is incorrect since the ground-coil does not pull the EWT all the way to the soil temperature. This point is made clearer in Figure 6. In cooling, an ASHP pumps heat from a low temperature source, the home, to a higher temperature sink, the outdoor air. Since heat normally would not flow in this direction, work input is required in the compressor, and this work input decreases as the outdoor temperature decreases. A GCHP, on the other hand, pumps heat from the home to the water at temperature EWT, which in turn rejects its heat to the soil. Thus, a much more valid comparison to make is between the EWT and the outdoor air.

Figure 7 presents hourly average values of EWT and outdoor temperature over the 1990 cooling season. This plot shows that at no point during the day, on average, was the EWT below the outdoor air temperature. EWT came closest to the outdoor air temperature from about noon to 7 P.M., when it was within 3°C. Also included in the figure is the undisturbed soil temperature at depths below 12 m, which was nearly always lower than the outdoor air temperature. This figure clearly demonstrates the problem of justifying GCHP use based solely on comparing outdoor air temperature to soil temperature. The soil temperature was nearly always cooler than the air temperature and the EWT was never cooler than the air temperature. The average value of EWT for this data was 33.1°C, which was 4.7°C warmer than the average outdoor air temperature of 28.4°C.

While comparing EWT and outdoor air temperature is more correct than comparing soil temperature and outdoor air temperature, it tacitly assumes that the condenser of a GCHP and an ASHP have the same effectiveness. This is not true, because water is a substantially better heat transfer medium than air. Since the goal of lowering the temperature of the heat sink is to lower the refrigerant condensing temperature (or pressure), a higher value of EWT may still result in a lower condensing temperature than a somewhat lower outdoor air temperature. Thus, there exists some advantage which a GCHP holds over an ASHP because it allows the refrigerant in the condenser to more closely approach the temperature of the heat sink. Determining the magnitude of this advantage definitively would require monitoring the refrigerant conditions, which was not done. As an approximation, engineering representatives of the GCHP vendor were asked to provide estimates of the temperature difference between the enteringwater and the saturated refrigerant in the condenser. Their estimate was 2.8 to 5.6° C. Data from a well instrumented ASHP [Farzad, 1990] indicated temperature differences of 10.6 and 11.1°C at outdoor temperatures of 27.8 and 35°C, respectively. Thus, a GCHP whose EWT is 5 to 8.3°C higher than the outdoor air temperature may still produce the same condensing temperature as an ASHP operating in the same environment. This is only an estimate, since the ΔT's for both the GCHP and the ASHP are dependent on the size and/or heat conductance of their respective condensers, as well as operating conditions. Still, this suggests that the GCHP from this project may have operated, on the average, with a slightly lower condensing temperature (0.8 to 3.6°C) than an ASHP would have under the same conditions. This is despite the fact that its EWT was an average of 4.7°C higher than the outdoor air temperature.

Seasonal Performance

The bottom line description of heat pump performance is normally a seasonal efficiency. Seasonal efficiencies have been determined for the 1990, 1991, and 1992 cooling seasons and the 1990/91 and 1991/92 heating seasons. The measure of seasonal cooling performance is the Seasonal Energy Efficiency Ratio (SEER), defined as:

$$SEER = \frac{Energy\ Removal\ in\ Btu}{Total\ Energy\ Input\ in\ W} \quad (5)$$

This is equivalent to the SEER used in the DOE rating procedure [DOE, 1986] in that it includes transient effects. For heating, the seasonal performance measure that will be used is the Heating Coefficient of Performance (HCOP), defined as:

$$HCOP = \frac{Energy\ Added\ in\ Btu}{Total\ Energy\ Input\ in\ Btu} \quad (6)$$

The SEER is dimensionally incorrect, and for this reason is 3.412 times higher than a seasonal COP for cooling. It is used herein since it is the standard reference for ASHPs, however.

The SEER values for the three cooling seasons are summarized in Table 2. The first column in Table 2 includes the raw SEER values of 10.8, 10.2, and 9.7. These values are comparable to middle efficiency ASHPs. The second column of Table 2 lists the indoor conditions that were maintained for each of the three cooling seasons. The average indoor air temperatures of 21.2°C, 20.6°C, and 20.4°C are well below the 26.7°C temperature used in the DOE rating procedure. The third column in Table 2 is the estimated SEER that would have been achieved if the conditioned space had been maintained at the DOE rating conditions. These values were estimated using the regression equations that described the GCHP performance, simply replacing the value of return air enthalpy that corresponded to the measured conditions with that from the DOE rating procedure. They are conservative estimates, because if the unit had been maintained at the warmer conditions from the DOE rating procedure the average EWT value would have also been lower. In terms of comparisons with ASHPs, however, these conservative SEER estimates are comparable to the most efficient single-speed ASHPs.

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Table 2 - Seasonal Cooling Performance for the Experimental GCHP

Cooling Season	Measured SEER	Average Return Air Temperature (°C)	Corrected SEER
1990	10.8	21.2	12.4
1991	10.2	20.6	11.7
1992	9.7	20.4	11.2

An interesting point from Table 2 is that the SEER decreased for each of the cooling seasons over which it was monitored. Two factors are believed to have contributed to this decrease. First, the conditioned space was maintained at cooler temperatures each year, as indicated in the table. Second, a substantial blockage of the return air duct was discovered when the experimental equipment were removed. This resulted in decreased airflow for the later cooling seasons, which would also tend to decrease the efficiency of the unit.

The estimated HCOPs for the 1990-91 and 1991-92 cooling seasons are listed in Table 3. These HCOPs were 3.5 and 3.1, respectively, and are comparable to the highest efficiency variable speed ASHPs that are currently available.

Table 3 - Seasonal Heating Performance for the Experimental GCHP

Heating Season	HCOP
1990-91	3.5
1991-92	3.1

SUMMARY

A nominal 10.6 kW (3 ton), vertical U-tube configuration GCHP was installed in Abilene, TX and monitored over three cooling seasons and two heating seasons. Transient performance data indicated that the EWT was most favorable early in the on-cycle due to heat transfer with the ambient soil during the off-cycle. This behavior was more pronounced for longer off-times. This resulted in increased efficiency early in the cycle, which is contrary to the typical behavior of ASHPs. This trend was observed in both cooling and heating.

Comparisons of the average values of EWT and outdoor air temperature during the cooling season indicated that the EWT was always warmer than the outdoor air. It was closest to the outdoor air temperature from noon to 7 P.M., when the two temperatures were within about 2.8°C. Since the water to refrigerant condenser of the ASHP was more effective than an air cooled condenser from an ASHP, though, it is likely that the average condensing temperature of the GCHP was lower than an ASHP would have achieved at similar conditions.

The SEER of the monitored system was comparable to middle of the line ASHPs. This was largely due to the conditioned space being maintained at temperatures well below those at which ASHPs are rated, however. If corrected for this difference, the SEERs were comparable to those of high efficiency single-speed ASHPs. The HCOPs of the GCHP were higher than those for even the highest efficiency variable-speed ASHPs.

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NOMENCLATURE

COP	Coefficient Performance
COP^+	Nondimensional COP based on Steady State EWT
c_p	Specific heat of air (kJ/kg·K)
DT	Temperature Difference (°C)
EWT	Entering Water Temperature (°C)
HCOP	Heating Coefficient of Performance
i_r	Return Air Enthalpy (kJ/kg)
i_s	Supply Air Enthalpy (kJ/kg)
$\dot{P}_{comp}$	Compressor Power (kW)
$\dot{P}_f$	Fan Power (kW)
$\dot{P}_p$	Pump Power (kW)
$\dot{P}_{sh}$	Strip heat Power (kW)
$\dot{P}_t$	Total Power (kW)
Q	Volumetric Flow Rate (m^3/s)
Q_c	Cooling Capacity (kW)
Q_{cond}	Condenser Capacity (kW)
Q_{gc}	Ground-Coil Capacity (kW)
r	Air Density (kg/m^3)
SEER	Seasonal Energy Efficiency Ratio

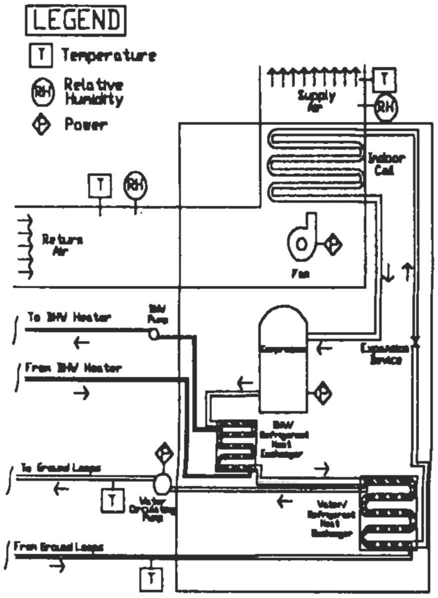

Figure 1 - Schematic of GCHP and Instrumentation

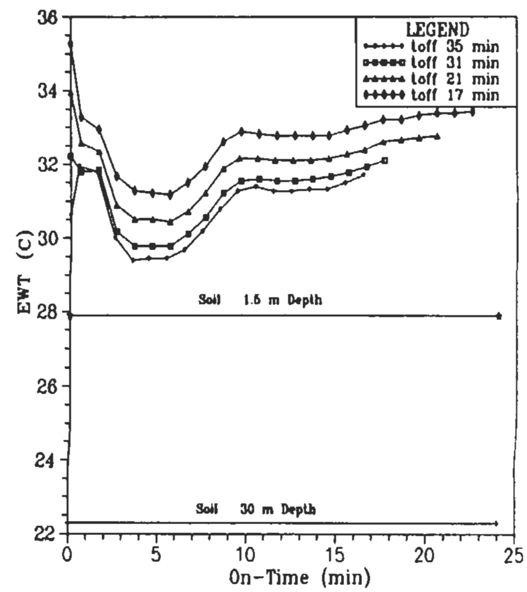

Figure 3 - Effect of On-Time and Off-Time on EWT during Cooling

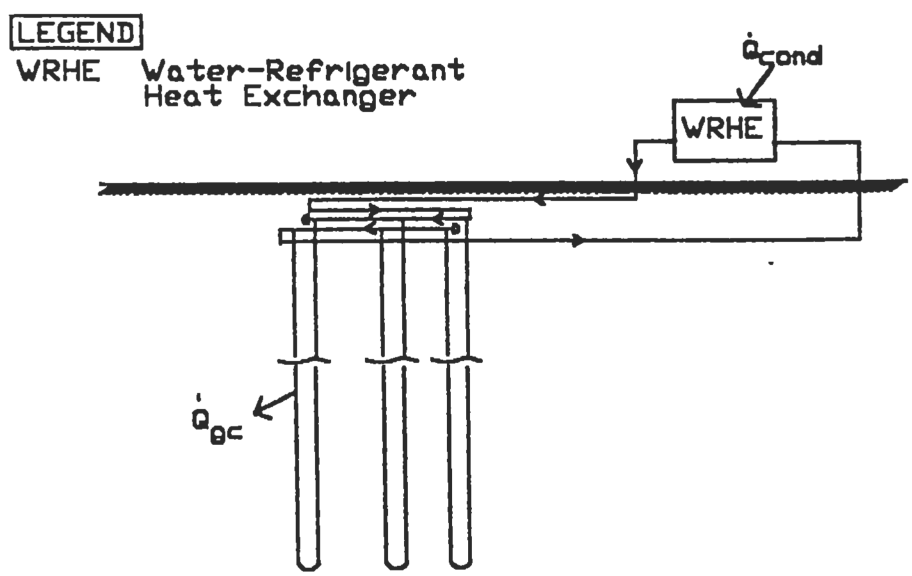

Figure 2 - Schematic of Ground-Coil Flow Configuration

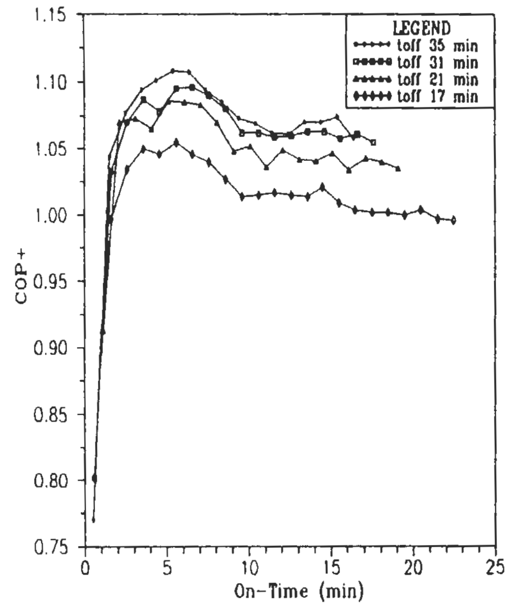

Figure 4 - Effect of On-Time and Off-Time on COP+

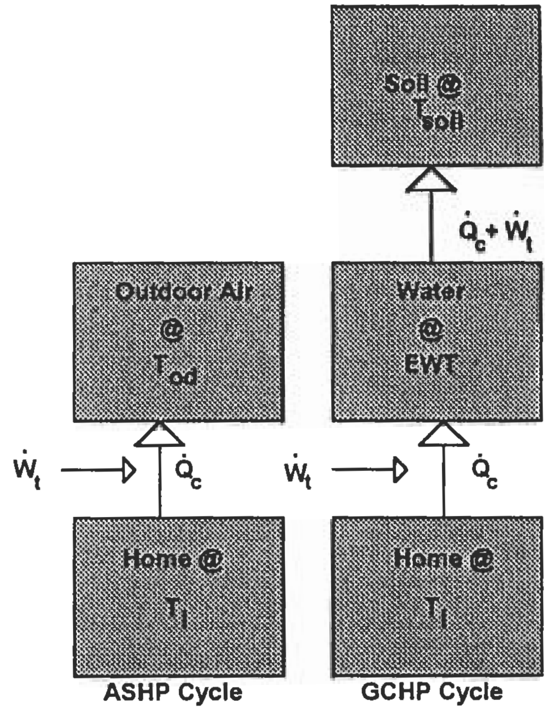

Figure 6 - Simple Schematic of ASHP and GCHP Cycles

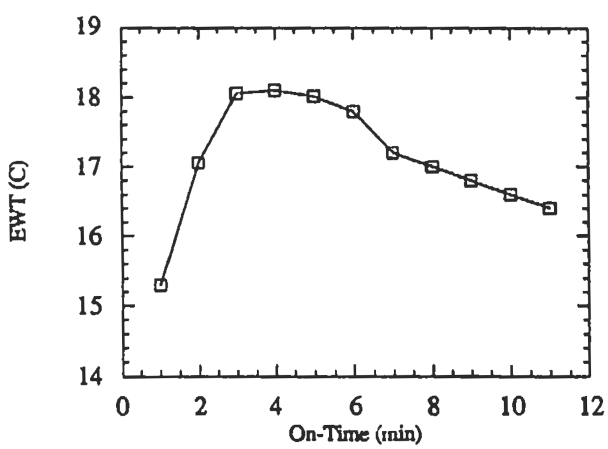

Figure 5 - Variation of EWT with On-Time during Heating Operation

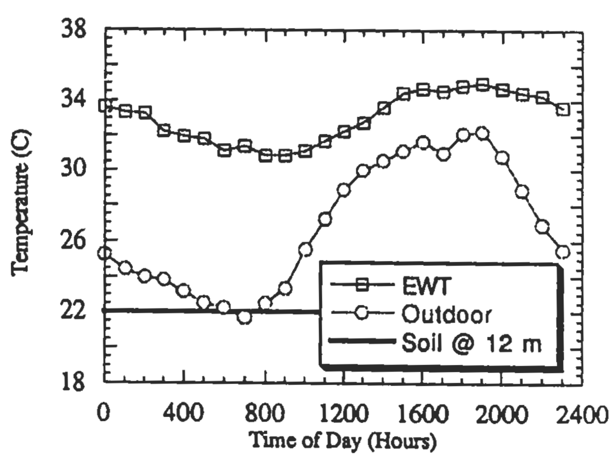

Figure 7 - Comparison of Hourly Average Values of EWT and Outdoor Air Temperature